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Effect of inlet cooling on the performances of isothermal main air compressors used for ASU applications

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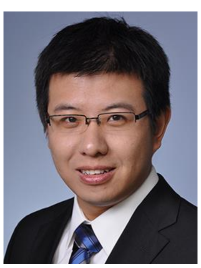

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ABSTRACT

Main air compressors used for air separation units (ASU) applications have a suction at ambient conditions and deliver air to a pressure range between 5.6 ~ 6.5 bara. Therefore, the performances of the compressor are greatly affected by the seasonal variation of ambient conditions (winter/summer conditions). Since the compressor must be sized for the maximum volume flow i.e. in the "summer" conditions, it results that when the seasonal difference is significant (i.e. >15°C) casing, stages as well as the cooler design must be oversized. Therefore, during "average day" the compressor runs always at off design condition, hence compromising efficiency.

One method to mitigate the impact of such large variations of ambient conditions is to have an inlet cooler which can operate to keep the inlet temperature (and therefore volume flow) below a certain limit during summer.

In this paper, a study is presented where different types of cooling technologies are applied in inline isotherm (intercooled) compressors. In particular, a "passive" and "active" cooling method is described and investigated. The first method does not require additional power (except for pumping the water in the inlet cooler, but the effort is minimal) but the cooling effect is dependent on ambient conditions. Consequently, the inlet air can only be chilled to wet bulb temperature. The second method overcomes the limitation of "passive" cooling but requires significant power consumption to drive the refrigeration (mechanical or adsorption) chiller.

The result is that the "passive" cooling method is most attractive and is most beneficial since it requires nearly no extra power and results in an optimal compressor design. The calculated benefit is up to 1% in total compression power compared to a conventional system without inlet cooling. This system has also the advantage of being very flexible since the inlet cooling can be switched on only during summer and left with no influence during other periods of the year.

INTRODUCTION

Main air compressors (MAC) and booster air compressors (BAC) are nowadays commonly used in air separation unit (ASU) business. These machines have the purpose to compress air from ambient conditions to different pressure levels and deliver it to the cold box for refrigeration and rectification so that different gases which comprised air (N2, O2, Argon and others) can be separated. Such a compression process is responsible for more than 60% of the total power required by an ASU. A very efficient way to reduce such high demand of compression power (as more as higher is the pressure ratio) is to implement cooling after each compression stage.

In the last 100 years different technical solutions have been applied in order to optimize the inter-stage cooling process, however, nowadays in the market there are mainly 2 different types of isotherm compressors: Inline compressors (with integrated coolers) and geared-type compressors (with external coolers). The first type (shown in Figure 1 and called "RIKT" compressor) is used as MAC. It operates typically with constant speed (electro motor or steam turbine driven) and it is regulated, in flow, by adjustable inlet guide vanes (IGVs).

MAC is designed to have a suction at atmospheric conditions and compress air typically until 5.7 ~ 6.5 bar, depending on the process downstream. Such machines are always driven either by an electric motor (until power of about 40 MW) or by a steam turbine (for higher power) which usually drives also a BAC on the other shaft end. The rotational speed is normally constant, only a small variation is allowed in turbine driven units (not more than 2-3%). The flow range of the MAC must be as large as possible since the MAC regulates the whole process ASU downstream and, therefore, controls the production of different products (liquid/gaseous N2, liquid/gaseous O2, Argon etc.). Typically, the flow range is from 75% until 105% of the design mass flow. Such flow variation is achieved by installing adjustable guide vanes (IGVs) at the inlet of the compressor, which throttle the flow to the desired capacity. The maximum flow capacity of the compressor is achieved when the IGVs are turned about 10°-15° in the rotation direction, since this provides the highest relative speed compared with the 1st impeller motion. The optimum performances are achieved when the IGVs are aligned with the flow i.e. at 90°.

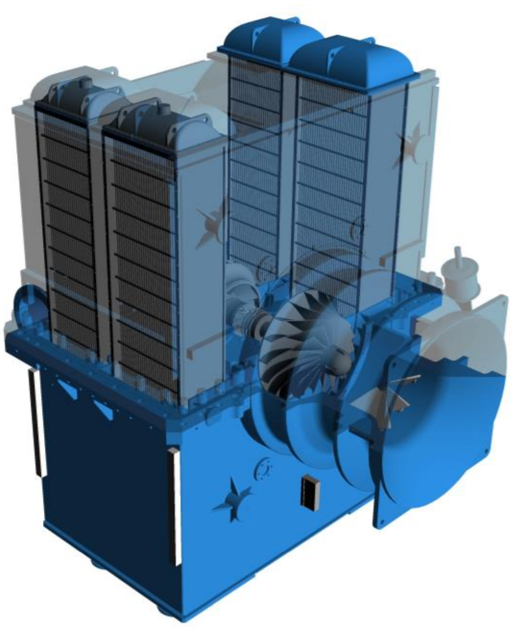

Figure 1: The inline isothermal compressor RIKT

The compressors are commonly sized according to the maximum volume flow (up to 110% load, related to the mass flow) at "summer" conditions, since the volume flow is proportional to the temperature. In these conditions the IGVs are at the maximum opening angle. At nominal design point (100% load), the compressors are supposed to reach maximum efficiency. The nominal design temperature is usually assigned as same as the "average day" temperature. However, the "average day" temperature is several degrees lower than the "summer" case. This implies that the IGVs must be notably closed (up to 55°-60°), and therefore, the compressor operates at a point far from its optimum. Consequently, any reduction of such seasonal variation on the volume flow is beneficial for the overall compressor performances. One method of reducing such variation at inlet conditions is to apply inlet cooling when the ambient temperature is high (e.g. > 35°C). Inlet cooling technologies are very well known in gas turbine business so that power output during summer is increased and efficiency gained [1]. However, it is very rarely applied in compressor technologies and it was never applied so far in RIKT type compressors for ASU application.

The effect of inlet cooling is to mitigate large differences of inlet conditions due to seasonal variation, and therefore, allow the compressor to operate on average at higher efficiency. By using an inlet cooler, two benefits appear: When the inlet temperature is almost constant between "summer" and "design day" 1) the design point can be achieved at optimal IGV position 2) stages and coolers selection can be optimized so that best efficiency at both "summer" and "average day" condition can be achieved and oversizing of the compressor can be avoided. The achieved savings in the total compressor power consumption is depending on a number of factors as compressor loading, differences between summer/winter conditions, humidity levels and investment costs. In the following, some thermo-economical cases are studied and an estimation of the power savings in one year of compressor operation is given.

INLET AIR COOLING METHODS

Various methods of inlet air cooling for improving the gas turbine performance by operating at higher ambient temperatures have been comprehensively investigated and described in open publications [1, 2 and 3]. Generally, available inlet air cooling methods can be classified into two groups: passive cooling and active cooling.

Passive Cooling

Passive cooling is based on evaporation of water in the inlet of the compressor. Due to evaporation, the inlet air is humidified and the latent heat of evaporation is absorbed from the inlet air. As a result, the inlet air is cooled. The effective cooling capacity is limited by the humidity, because the evaporation process only takes place as long as the air is not saturated (<100% relative humidity). Evaporative cooling can be considered as a nearly adiabatic process, because heat transfer between inlet air flow and its surrounding is negligible. At saturated state the inlet air can be cooled at minimum to the wet bulb temperature. In order to characterize the quality of the evaporative cooler, the effectiveness η_e is introduced, which is defined as the ratio of the temperature difference between inlet temperature T_∞ and outlet temperature T_1 of the cooler and the temperature difference between inlet and wet bulb temperature of inlet air T_{wb}.

$$\eta_e = \frac{T_\infty - T_1}{T_\infty - T_{wb}} \tag{1}$$

The traditional evaporative cooler with fill pack and the fogging system are common methods for passive cooling, which have been widely used because of the simplicity of components and its low investment cost. The fill pack of the evaporative cooler is made of fibrous corrugated material, such as glass fibres, impregnated paper or light metal. Water is distributed from the top of fill packs and evaporates with the air flow. The effectiveness of these coolers is normally about 85% - 93%, depending on the size of its available cooling surface. The operation of evaporative coolers requires an air flow velocity of 2 – 4.5 m/s. Above a velocity of 2.5 m/s the water droplets can detach from the surface of the cooler and be sucked into the compressor. Therefore, a droplet separator is required to avoid the erosion risk of the compressor impeller. The fill packs and water separator will cause a pressure drop of up to 500 Pa [11]. However, water quality requirements are less stringent than those required for fog-cooling systems.

Direct inlet fogging is a method of cooling where demineralized water is converted into a fog by means of special atomizing nozzles operating at 70-200 Bar. The nozzles create a large number of micron size droplets (size between 5-20µm), which will evaporate at the compressor inlet and cooling the inlet air to, minimum, wet bulb conditions. Pressure losses caused by fogging systems, in contrast to evaporative coolers, are much lower, typically less than 50 Pa. However, the pump of the fogging system has higher power consumption due to the high injection pressure.

Details pertaining to thermodynamics and practical aspects of fogging have been described in [5, 6, 7]. This technique allows close to 100% effectiveness in terms of attaining close to 100% relative humidity inlet and thereby gives the lowest possible temperature (the wet bulb temperature) without additional refrigeration as for active cooling. Several studies focused on the effects of such fogging technologies on gas turbine performances [8, 9, 10]. A picture showing a typical high pressure fogging nozzle array is shown in Figure 2.

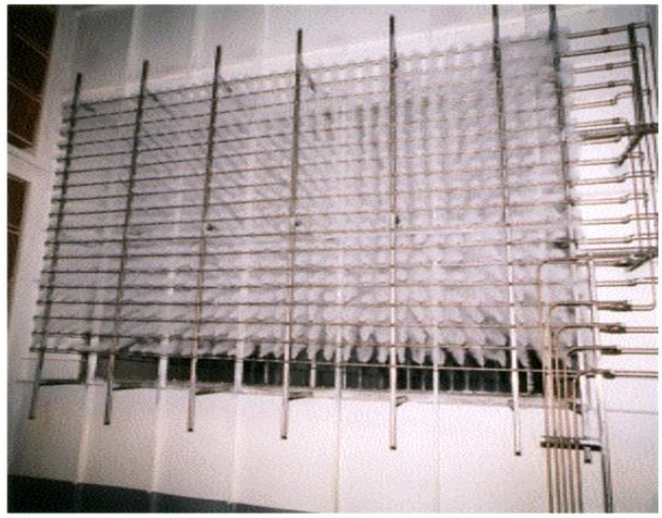

Figure 2: Fogging nozzles for typical gas turbine application

Inlet fogging includes a series of high pressure reciprocating pumps providing demineralized water to an array of fogging nozzles located after the air filter elements. For ensuring a safe operation, a certain safety distance (about 1-2%) to the saturation state is to hold. Regarding to the risk of clogging caused by depositing, the water muss be filtered and demineralized previously.

Active Cooling

To overcome the limitation of passive cooling, in which the inlet air can only be cooled to the wet bulb temperature, active cooling can be implemented. Active cooling requires external power to achieve the desired cooling temperature, therefore, it usually involves higher system complexity, space requirement, investment and operating costs than passive cooling. Despite these drawbacks, the active cooling provides also several advantages. Above all, the cooling effect is independent of weather conditions. Constant inlet conditions can be ensured, so that it allows an optimal and stable operation efficiency during the year. Mechanical and absorption refrigeration systems are commonly used techniques for active cooling.

Mechanical Refrigeration System

A mechanical refrigeration system uses a circulating refrigerant as a medium, which absorbs and removes heat from the inlet air by means of a heat exchanger and subsequently rejects that heat elsewhere. Typically, the evaporator is directly installed in the inlet of compressor as a heat exchanger and the inlet air can be cooled down to 3–4 °K higher than the refrigerant temperature. The refrigerant vapour is compressed by using a centrifugal, screw, or reciprocating compressor, which are mostly driven by electric motor. Consequently, the electrical power consumption of the mechanical refrigeration system is high. However, the mechanical refrigeration system has a high coefficient of performance (COP), which can be up to 5.

Absorption Refrigeration System

The absorption refrigeration system utilizes waste heat instead of electricity as energy source. This ability provides an energy savings opportunity if waste heat is available. In the absorption cycle, LiBr and water is the preferred refrigerant and in combination an absorbent agent due to their chemical stability and operational safety. A conventional system produces chilled water at temperatures up to 2°C as cooling media, so it is possible to use direct contact air-cooler to achieve a smaller temperature difference (about 2°C) between chilled water and cooled air, compared to the indirect contact air-cooler. Besides the conventional system various types of absorption cycles at different levels of system complexity and efficiency exist. A single stage system will have a COP of 0.7–0.8 and a double-effect unit a COP of 1.4. Unlike the mechanical refrigeration system the absorption refrigeration system does not lose efficiency at part load and provides higher operational flexibility. Absorption systems have typically higher investment costs and space requirements, but lower operating and maintenance costs than mechanical refrigeration systems.

MODELING OF INLET AIR COOLING SYSTEMS

Thermodynamic Model

Passive Cooling

By using passive cooling the inlet air temperature after cooling is limited to wet bulb temperature. Assuming that the effectiveness of cooler is 100% and the temperature of evaporated water $\mathrm{T_w}$ is equal to air temperature at outlet of the cooler, the wet bulb temperature is calculated as follows:

According to conservation of mass:

$$\mathrm{X}_\infty \cdot \dot{\mathrm{m}}_\mathrm{a} + \dot{\mathrm{m}}_\mathrm{w} = X_s(T_{wb}) \cdot \dot{\mathrm{m}}_a \qquad (2)$$

where X_∞: specific humidity of air before cooling, X_s: specific humidity of air at saturated state after cooling, $\dot{\mathrm{m}}_\mathrm{a}$: mass flow of dry air, $\dot{\mathrm{m}}_\mathrm{w}$: mass flow of water, T_wb: Wet bulb temperature.

According to conservation of energy:

$$\dot{\mathrm{m}}_\mathrm{a} \cdot \mathrm{h}_{\mathrm{l+x},\infty} + \dot{\mathrm{m}}_\mathrm{w} \cdot h_w = \dot{\mathrm{m}}_a \cdot h_{l+x,wb} \qquad (3)$$

where $\mathrm{h}_{\mathrm{l+x},\infty}$: enthalpy of air before cooling, h_w: enthalpy of water, $h_{\mathrm{l+x,wb}}$: enthalpy of air at saturated state after cooling.

Enthalpies in the equation written above are defined as:

$$\mathrm{h}_{\mathrm{l+x},\infty} = \left(\mathrm{c_{pa}} + \mathrm{c_{pv}} \cdot \mathrm{X}_\infty\right) \cdot \mathrm{T}_\infty + \mathrm{r}_0 \cdot \mathrm{X}_\infty \qquad (4)$$

$$h_w = \mathrm{c_{pw}} \cdot T_{wb} \qquad (5)$$

$$h_{l+x,wb} = \left(\mathrm{c_{pa}} + \mathrm{c_{pv}} \cdot X_s(T_{wb})\right) \cdot T_{wb} + \mathrm{r}_0 \cdot X_s(T_{wb}) \qquad (6)$$

where $\mathrm{c_{pa}}$: specific heat capacity of dry air, $\mathrm{c_{pv}}$: specific heat capacity of water vapour, $\mathrm{c_{pw}}$: specific heat capacity of water, r_0: latent heat of evaporation of water, T_∞: Temperature of air before cooling.

Substituting Equation (2) into Equation (3), it results:

$$\mathrm{h}_{\mathrm{l+x},\infty} + (X_s(T_{wb}) - \mathrm{X}_\infty)h_w = h_{l+x,wb} \qquad (7)$$

Substituting Equations (4),(5) and (6) into (7) Equation, it results:

$$T_{wb} = \frac{\mathrm{r}_0\left(\mathrm{X}_\infty - X_s(T_{wb})\right) + \left(\mathrm{c_{pa}} + \mathrm{c_{pv}} \cdot \mathrm{X}_\infty\right)}{\left(\mathrm{c_{pv}} - \mathrm{c_{pw}}\right) \cdot X_s(T_{wb}) + \mathrm{c_{pw}} \cdot \mathrm{X}_\infty + \mathrm{c_{pa}}} \qquad (8)$$

where $X_\mathrm{s}(T_\mathrm{wb})$ is a function of saturated vapor pressure $p_\mathrm{s}(T_\mathrm{wb})$:

$$X_s(T_{wb}) = 0{,}622\,\frac{p_s(T_{wb})}{\mathrm{p} - p_s(T_{wb})} \qquad (9)$$

where saturated vapour pressure $p_s(T_\mathrm{wb})$ is defined as [12]:

$$P_s(T_{wb})[bara] = 10^{-3} \cdot 10^{0{,}83246 - \frac{1799{,}73}{T_{wb}[°\mathrm{C}] + 238{,}734}} \qquad (10)$$

Thus, the coupled equation (8) $T_\mathrm{wb} = f\left(X_s(T_\mathrm{wb})\right)$ can be solved with an iterative method.

Instead of calculation, wet bulb temperature can be also determined based on the psychrometric chart. Figure 3 illustrates the path that air undergoes a change from ambient state (a) to the cooled state (b). With the approximation that the enthalpy before and after the passive cooling remains constant, i.e. $\mathrm{h}_{\mathrm{l+x},\infty} = \mathrm{h}_{\mathrm{l+x,wb}}$ (because $\dot{\mathrm{m}}_\mathrm{W}$ as well as $\dot{\mathrm{m}}_\mathrm{W} \cdot \mathrm{h_w}$ is negligibly small compared to $\dot{\mathrm{m}}_\mathrm{L} \cdot \mathrm{h}_{\mathrm{l+x},\infty}$), the adiabatic passive cooling process runs along with the isenthalpic line until reaching the saturated vapour line. The temperature at the point, in which the isenthalpic line and the saturated vapour line meet, is the wet bulb temperature. As an example the temperature and the relative humidity of the ambient state are assumed at 25°C and 50% and at the saturation state a cooled air temperature of 17.8 °C is attained.

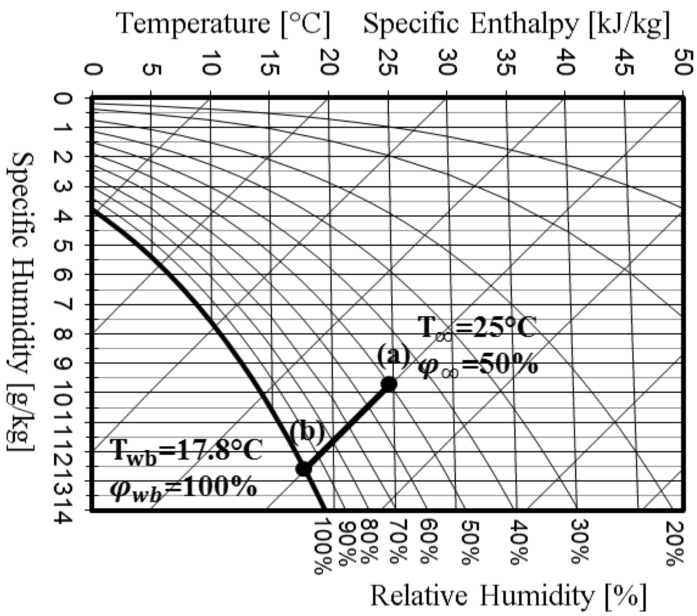

Figure 3: Representation of passive cooling process on psychrometric chart, (a): ambient state, (b): saturated state after passive cooling [16]

In this work, evaporative cooling was chosen and the effectiveness η_e of the cooler is given by manufacturer about 92%. Referring to the effectiveness, the compressor inlet air temperature after cooling and the temperature drop ratio at different ambient conditions can be obtained by using Eq. (1) and seen in Figure 4.

Passive cooling with water evaporation leads on one hand to water consumption and on the other hand to a mass flow increase in the compressor. These effects also have to be taken into account for further investigations. The increased mass flow can be calculated by using:

$$\dot{\mathrm{m}}_{\mathbf{w}} = (\boldsymbol{X}_{\mathbf{1}}(\boldsymbol{T}_{\boldsymbol{wb}}) - \mathbf{X}_{\infty}) \cdot \dot{\mathrm{m}}_{\boldsymbol{a}} \qquad (11)$$

where $\mathrm{X}_1(\mathrm{T}_\infty)$ is given by:

$$X_1(T_{wb}) = 0{,}622 \frac{\eta_e p_s(T_{wb})}{\mathrm{p} - \eta_e p_s(T_{wb})} \qquad (12)$$

The water consumption rate (i.e. the mass flow) increases in the compressor related to dry air mass flow at different ambient conditions, which can be seen in Figure 5.

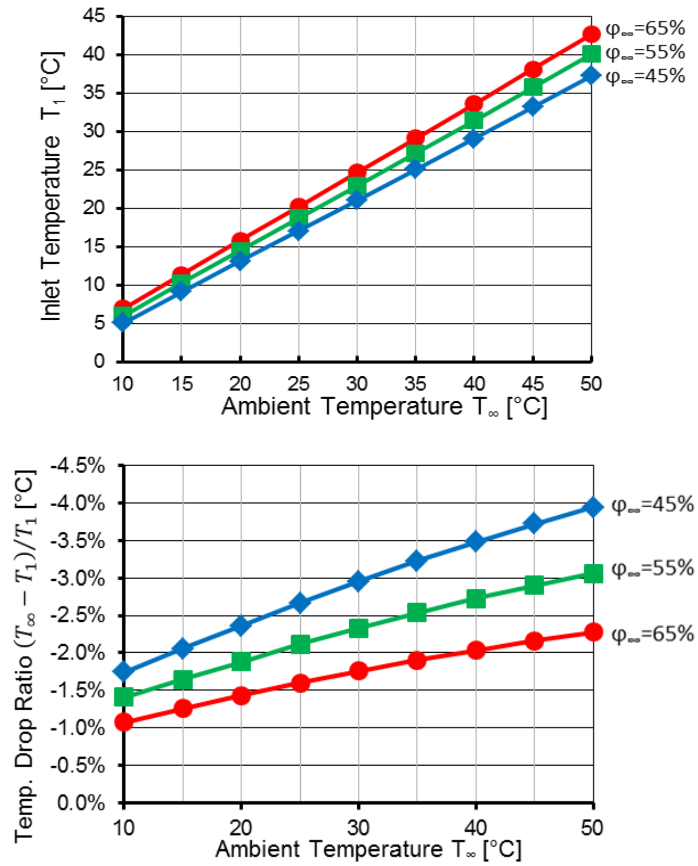

Figure 4: Compressor inlet air temperature after cooling (top) and temperature drop ratio (bottom) as a function of ambient temperature

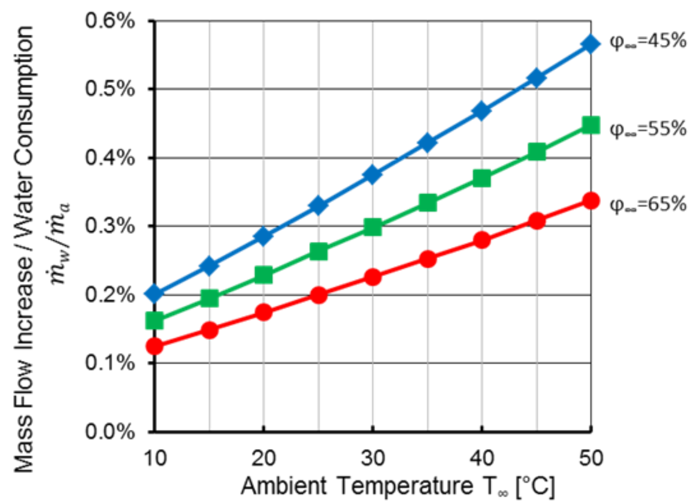

Figure 5: Water consumption rate, i.e., mass flow increase in the compressor at different ambient conditions

Active Cooling

With active cooling, the inlet air can be cooled under the wet bulb temperature. The path of the active cooling process can be seen on the psychometric chart as it undergoes a change from assumed ambient state (a) to the desired cooled state (c) with temperature of 5°C. Depending on the different types of coolers the path can be also different before reaching the saturated state, (b) for direct contact cooler and (b') for indirect contact cooler, respectively.

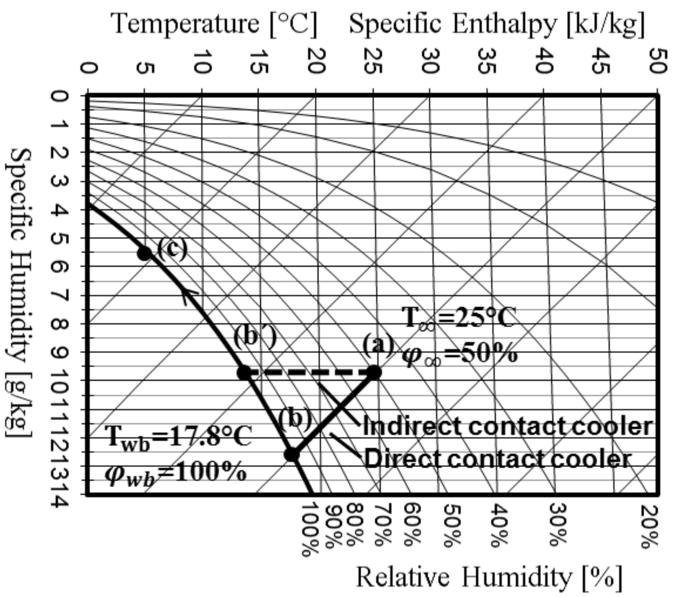

Figure 6: Representation of active cooling process on psychrometric chart, (a): ambient state, (b) saturated state for direct contact cooler, (b'): saturated state for indirect contact cooler, (c) desired cooled state [16]

For any further cooling from the saturated state, the water vapour starts to condensate and releases the latent heat. Thus, the cooling load rises dramatically due to the removal of latent heat during condensation. By considering the enthalpy changes at two different coolers in Figure 6, it can be seen, that the required cooling load remains the same, despite different paths on the psychrometric chart. In order to obtain the driving power for active cooling, the required cooling load has to be determined. The required cooling is defined as the total amount of heat to be removed from the air, i.e., enthalpy change between inlet and outlet of the cooler. The total cooling load is calculated by:

$$\begin{aligned}\dot{Q}_{tc} &= h_{1+X,\infty} - h_{1+x,c} \\ &= c_{pa}(X_\infty - X_c) + c_{pv}(X_\infty \cdot T_\infty - X_c \cdot T_c) \\ &\quad + r_0(X_\infty - X_c)\end{aligned} \tag{13}$$

where T_c: desired cooling temperature, X_c: specific humidity of desired cooling temperature.

As an example the cooling load for cooling the air from a temperature of 35 °C to diverse temperatures until 5°C at different relative humidity is shown in Figure 7.

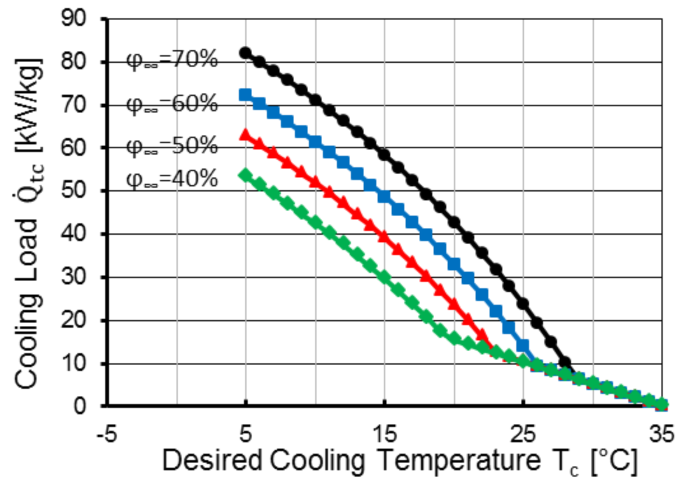

Figure 7: Cooling load for cooling the air from 35 °C to diverse temperature until 5 °C at different relative humidities

The driving power required to run the active cooling system can be determined by:

$$P_c = \frac{\dot{Q}_{tc}}{COP} \tag{14}$$

In the present analysis only a mechanical refrigeration system is considered in case of active cooling, the COP of mechanical refrigeration system is assumed about 5.

In contrast to passive cooling, the mass flow of the inlet air decreases after reaching the saturated state because of the water condensation. The decreased mass flow can be calculated according to the equation (15).

$$\dot{m}_w = (X_s(T_c) - X_\infty) \cdot \dot{m}_a \tag{15}$$

In Figure 8, the mass flow reduction of inlet air at 40°C and at different relative humidities is shown.

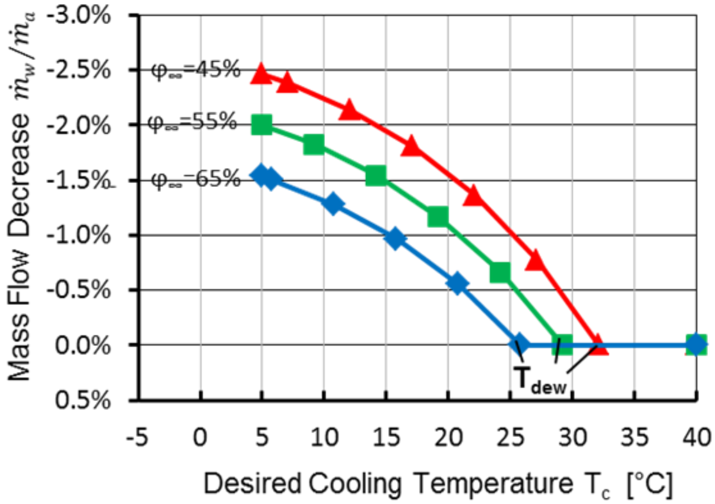

Figure 8: Mass flow decrease in the compressor at different ambient conditions

Summer		
Cases	"S" and "M"	"L"
No cooling	T = 35 °C φ = 65.5%	39.7 °C 54%
Active cooling (original design)	T = 27.6 to 5 °C φ = 100%	28.6 to 10 °C 100%
Active cooling (optimized design)	T = 27.6 to 5 °C φ = 100%	28.6 to 5 °C 100%

Average day		
Cases	"S" and "M"	"L"
No cooling	T = 25 °C φ = 65.5%	21.1 °C 54%
Active cooling (original design)	T = 20 to 5 °C φ = 100%	15 to 10 °C 100%
Active cooling (optimized design)	T = 20 to 5 °C φ = 100%	15 to 10 °C 100%

Table 1: Inlet air conditions of investigated operation points at different desired cooling temperatures on "summer" and "average day" – Active cooling

TEST CASES ANALYSIS AND THERMO/ECONOMIC ASPECTS

In the following two sections, the results of the impact by applying active and passive cooling are presented. The calculations have been carried out by considering different types of rotors - standard and customized rotors. Standard rotors are designed according to MAN Diesel & Turbo design philosophy in order to have the best compromise between efficiency and cost in a wide operational range. Customized rotors are only designed to achieve the best efficiency in that specified operation points, by using customized impeller sizes, types etc. The effect of inlet cooling is not the same for standard and customized rotors since the stage performance matching is affected by different inlet volume flows. With a reduced volume flow the impellers can be downsized or the speed can be reduced. Since the intercoolers have a very high cooling capacity, the outlet temperature after the intercoolers is generally insensitive to the inlet temperature. This means that mainly the first stage of the compressor benefits from the inlet cooling. This is significant since the first stage requires about 40% of the total compression power.

Active Cooling

In this section, the power savings by applying active cooling at different desired cooling temperatures (up to 5 °C) as well as different compressor sizes have been investigated. The inlet air conditions of investigated operation points at different desired cooling temperatures for "summer" and "design day" operations and for different compressor sizes ("Small", "Medium" and "Large") are shown in Table 1. The compressor "S" and "M" are both configured with four stages and each has two and three intercoolers, respectively. The compressor "L" has three stages with two intercoolers.

Figure 9 and 10 present the power savings by applying active cooling for different compressor sizes at "summer" and "average day" operations as a function of the desired inlet temperature. Continuous lines represent standard and customized rotors layouts done at "summer" conditions while dashed lines represents optimized layouts done in case of active cooling at dew point.

It can be observed that by cooling down to the dew point the power savings in summer is ca. 0.7% (four-stage) to 1% (three-stage) by every 5°K temperature drop. This is a typical situation when a passive cooling device would be applied i.e. minimum inlet temperature achievable is the bulk temperature as explained in previous sections. By continuing cooling down to below the dew point (active cooling) the moisture contained in the inlet air condenses out and it causes a decrease of mass flow; therefore, additional power savings of ca. 0.8%. The power savings increase to about 1.5% (four-stage) and 1.7% (three-stage) every 5°K of temperature reduction. By further cooling, the operation point moves far away from the design point, so that the efficiency losses are getting larger i.e. the power savings curve becomes more flat (about 0.5% per 5 °C until the desired cooling temperature of 5°C). With optimized designs at each desired cooling temperature a nearly linear path with constant power savings of ca. 1.5% per 5°K temperature drop can be observed.

At the "average day" operation similar statements can be made. However, the power savings are much less than summer operation (even with optimized designs), because the compressor runs with further reduced mass flow, i.e. less moisture at same desired cooling temperature compared to

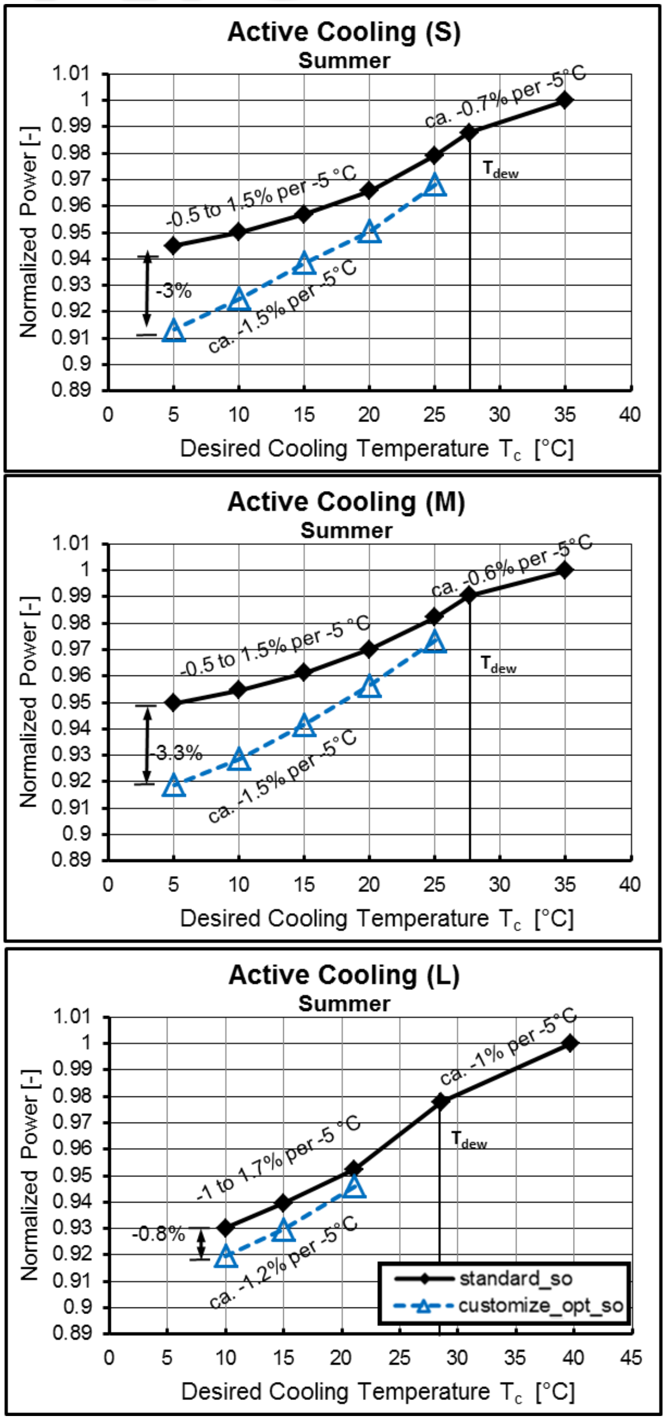

Figure 9: Comparison of the power savings by applying active cooling for different compressor sizes at “summer” operation (“so” stands for “summer” operation)

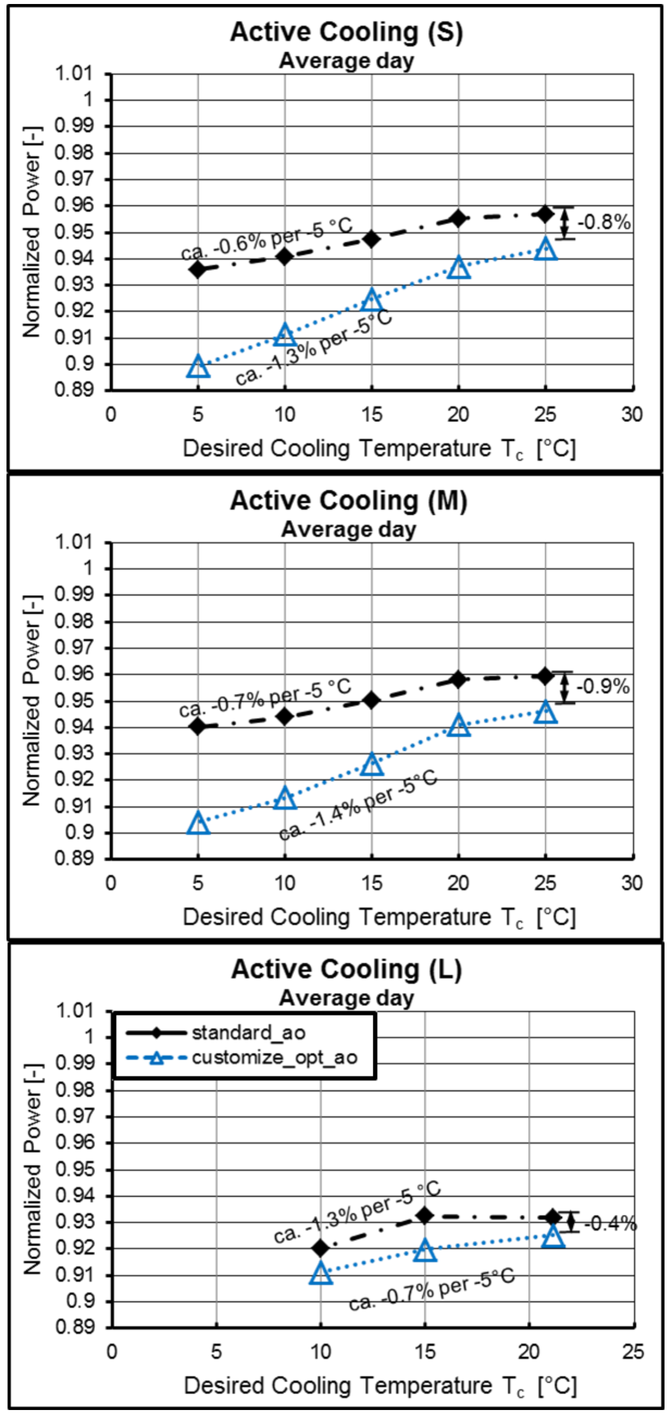

Figure 10: Comparison of the power savings by applying active cooling for different compressor sizes at “average day“ operation (“ao” stands for “average day” operation)

“summer” operation. On the one hand, the efficiency losses become larger and on the other hand the power savings regarding to moisture condensation become smaller.

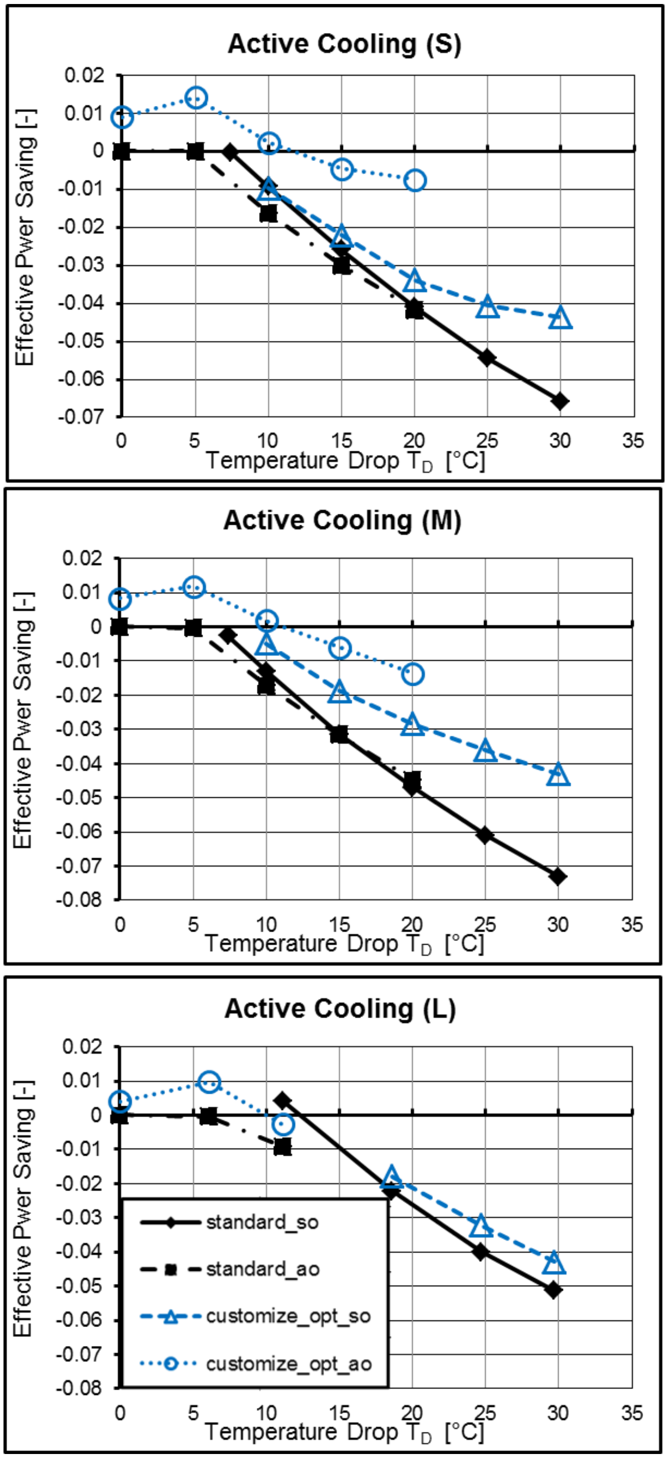

Figure 11: Effective power saving by applying active cooling

In Figure 11, the effective power savings of active cooling are shown. Such effective power savings take into account the actual compressor shaft power savings and the cooling effort required to achieve the desired inlet temperature. In case of active cooling, the cooling power required to achieve such inlet temperature is much greater than passive cooling, whereby the system will only require energy to pump the cooling water available through the spray nozzles. It can be seen from Figure 11 that only with the optimized design positive power savings of about. 1.5% can be attained at “average day” operation, of which 1 % is directly from the efficiency increase due to the optimized design. It implies that active cooling can only provide the rest of 0.5% power savings with a maximum available temperature drop of ca. 5°K. However, this can be better achieved by applying passive cooling with less expense.

Passive Cooling

Since the cooling effect of passive cooling (maximum temperature drop) substantially depends on the humidity of the inlet air, the power savings at different relative humidities (45%, 55% and 65%) have been investigated. The optimized designs have been carried out at operation point with cooled state (29.6 °C and 100% relative humidity) of inlet air of summer condition (35°C and 65% relative humidity). The inlet air conditions of investigated operation points at different relative humidities for different compressor sizes are shown in Table 2. The power savings by applying the passive cooling for different relative humidity are presented in Figure 12, with respect to 100% compressor load.

The results show the same trend on the different sizes of compressor. In summer, the power savings of three-stage and the four-stage compressors are between ca. 0.5% and 0.8%, which is almost independent of the humidity. One reason is that the increased cooling effect at lower relative humidity is partly compensated by the increased mass flow due to a higher amount of evaporated water. Also, this increased amount of water vapor has an indirect impact on the heat transfer of intercoolers.

The moisture contained in the air condenses and a condensate film is formed on the surface of the intercoolers. The lower the relative humidity of the air, the more water is evaporated, and thus, the more condensate is formed. This deteriorates the heat transfer at the cooler surface and causes the increase of inlet temperature of the compressor stage behind the intercooler. Furthermore, the larger the temperature drop, the higher the efficiency losses become. Overall, the benefit of larger temperature drops at lower relative humidity is completely neutralized by these factors above. In summer, the optimized designs can only provide slight efficiency improvements, i.e. power saving increase, of about 0.1%, because the operation points at different relative humidity are close to the design point with maximum efficiency.

Summer		
Cases	"S" and "M"	"L"
No cooling	T [°C] = 35 φ [%] = 45, 55, 65	T [°C] = 35 φ [%] = 45, 55, 65
Passive cooling (original design)	T [°C] = 26, 27.8, 29.6 φ [%] = 100	T [°C] = 26, 27.8, 29.6 φ [%] = 100
Passive cooling (optimized design)	T [°C] = 26, 27.8, 29.6 φ [%] = 100	T [°C] = 26, 27.8, 29.6 φ [%] = 100

Average day		
Cases	"S" and "M"	"L"
No cooling	T [°C] = 25 φ [%] = 45, 55, 65	T [°C] = 21.1 φ [%] = 45, 55, 65
Passive cooling (original design)	T [°C] = 17.7, 19.2, 20.6 φ [%] = 100	T [°C] = 14.5, 15.9, 17.1 φ [%] = 100
Passive cooling (optimized design)	T [°C] = 17.7, 19.2, 20.6 φ [%] = 100	T [°C] = 14.5, 15.9, 17.1 φ [%] = 100

Table 2: Inlet air conditions of investigated operation points at different desired cooling temperatures on "summer" and "average day" – Passive cooling

At "average day" operation, the compressor works with a much lower reduced volume flow as the part load, due to the lower temperature. By applying passive cooling at "average day" temperature, the additional volume flow decrease causes larger efficiency losses than at "summer" temperature, since the flow is much smaller, more unstable due to a reduced IGV angle. It can be seen that the power savings of passive cooling at "average day" operation provides insufficient benefit. Thus, the passive cooling must be shut down during "average day" operation.

Compared to "summer" operation, the benefit of the optimized designs at "average day" operation becomes more noticeable. With the optimized designs, the volume flow decrease at part load becomes smaller so that the IGV can be less closed than with original designs, and thus, a better efficiency of 0.5% can be achieved with the same decreased volume flow.

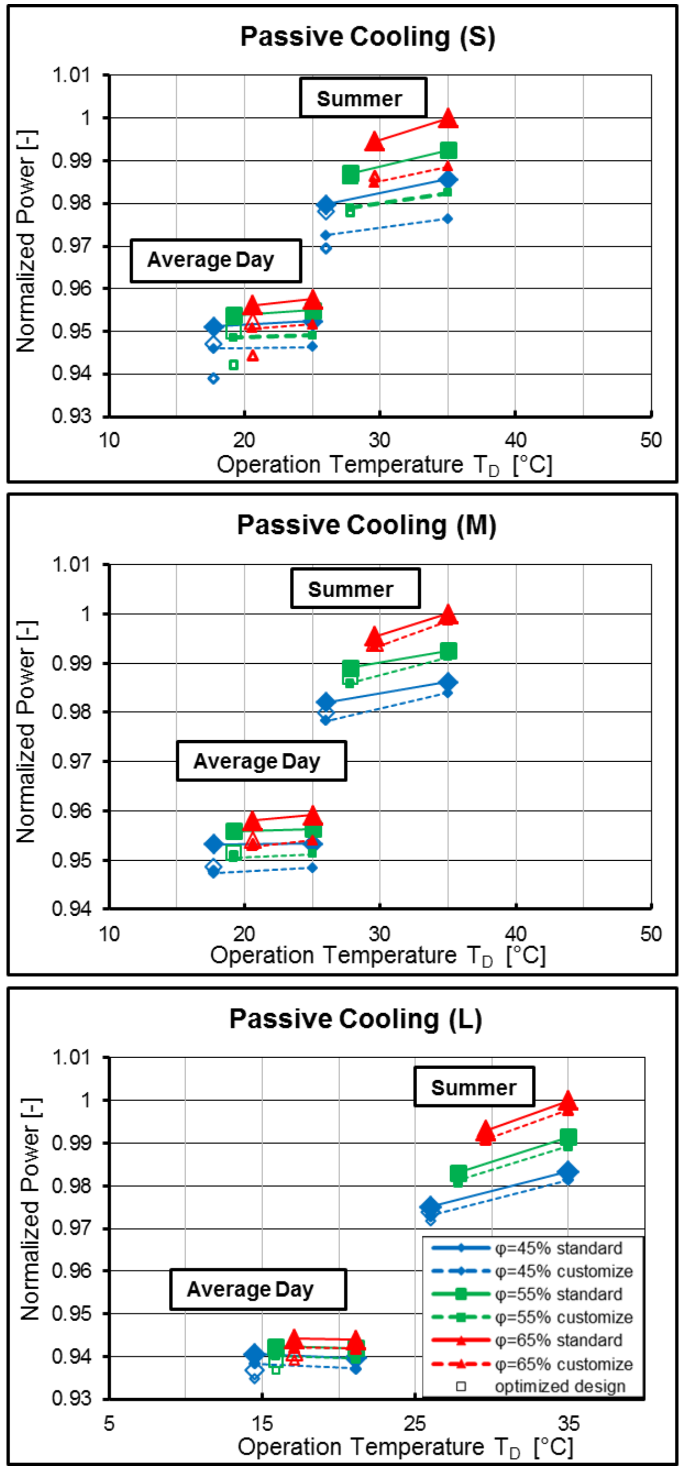

Figure 12: Power savings by applying power cooling at different relative humidity for different compressor sizes

THERMO-ECONOMIC ASPECTS

From the analysis of active and passive cooling described in the previous sections, it is concluded that the most attractive solution is the passive cooling, where additional energy required by the cooling system is minimal. Therefore, the cost/benefit analysis has been made only for this case based on a large 40MW MAC compressor. Two different industrial sites were considered where the difference in ambient temperature between summer and winter is substantial. In particular, Ashgabat (Turkmenistan) and Kashgar (China) were considered where a number of ASU plants are already in operation (Figure 12).

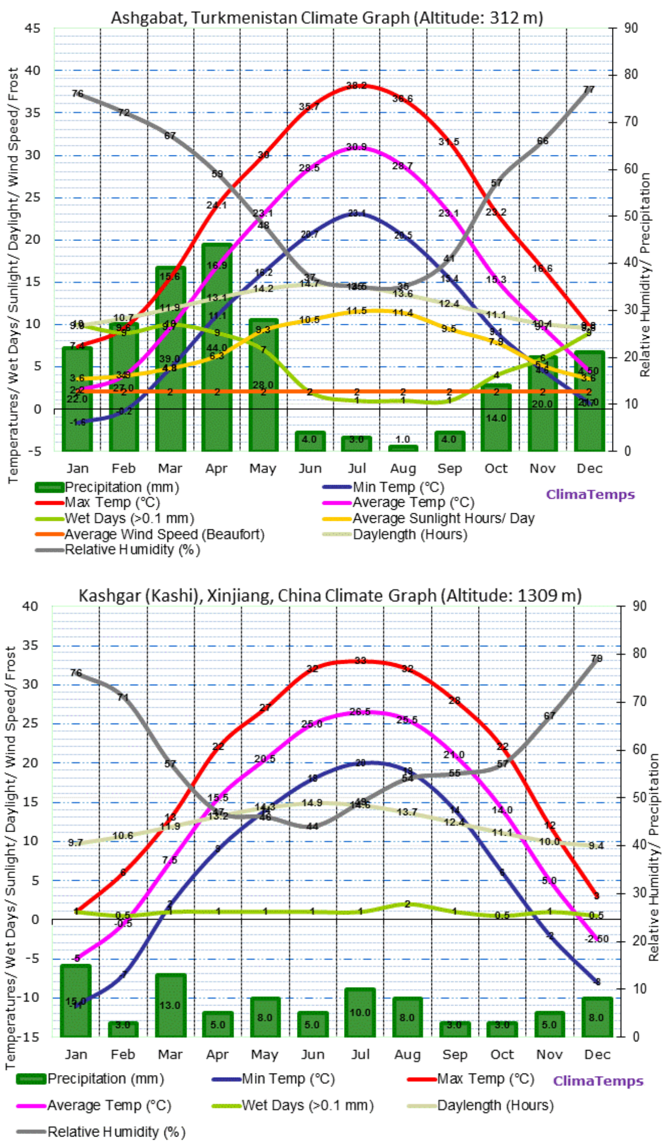

Figure 12: Weather conditions at Ashgabat (Turkmenistan) [17] and Kashgar (China) [18].

Ashgabat and Kashgar have quite high average temperature (pink curve) in summer (up to 38.2°C) with a low relative humidity and an “average day” temperature (i.e. “design” temperature) of about 25°C. Therefore, without cooling the RIKT compressor, it must be sized for a large volume flow, which would be reached for only 4 months during the year. It results that the maximum flow at “summer” conditions specified has to be calculated with an inlet temperature of 39°C. The “design” flow condition (at 25°C) is lower than 10% compared to the “summer” case. The RIKT compressor is then designed so that at “design” point the IGV must be closed up in the counter direction of the impeller rotation. Once the passive inlet cooling is applied, the minimum wet bulb temperature, which can be achieved is about 27°C (Figure 4) and this reduces the actual volume flow at “summer“ condition to an amount about 2.7% lower than in the previous case. By designing the compressor with such a low volume flow it results that at “design” and “summer” point there are power savings of about 0.7% by using standard rotors configuration. It can be observed that the absolute power savings with standardized impeller is larger than with the customized impeller, in case of optimized design. This is caused by restricted design options on the standardized impeller and may vary from case to case.

To determine the realistic power savings over 1 year operation, the compressor load profile should be considered. However, the compressor load depends on the consumer market, which varies from year to year. Therefore, instead of the compressor load profile, different annual average loads from 90% to 100% have been considered. During full load (100% load) operation, the passive cooling in Kashgar as well as in Ashgabat is only activated in July and from June to August, respectively. For operations below 90% load, the operation duration is reduced in to one month in Ashgabat, while in Kashgar it already had to be shut down.

The power savings at various annual average loads and the calculated total annual power savings are summarized in Table 3. The total power savings refer to a 24-hour operation with 335 operating days per year and they result from:

Overall power savings = Design (no cooler) + Summer (no cooler) + Summer (with cooler)

In this case, total power savings are composed of three parts: power savings by “average day” operation with deactivated cooler, power savings by “summer” operation with deactivated cooling by part load (90% load) and power savings by “summer” operation with activated cooler. The operation in these months with an average temperature higher than 27 °C is considered as “summer” operation. The remaining operating time is “average day” operation. The total duration of “summer” operation for Ashgabat as well as Kashgar is five and three months, respectively. By calculating the total power savings, it is assumed that, the power savings during the defined operating state remains constant.

It can also be noticed that the absolute power savings in machines with a standardized impeller is up to 20% larger than with a customized impeller. Since a higher efficiency can be achieved with the customized impeller so that the potential of efficiency improvement by a new optimized design is smaller than with the standard impeller.

Average Main Air Compressor load	90%	100 %
Duration of summer operation (Ashgabat) [months]	5	
Inlet cooling switched on (Ashgabat) [months]	1	3
Duration of summer operation (Ashgabat) [months]	3	
Inlet cooling switched on (Kashgar) [months]	0	1
Power savings using standardized rotors		
Power savings on summer operation [kW]	233	446
Power savings on average operation [kW]	349	309
Total power savings (Ashgabat) [10^6 kWh/y]	2,34	2,73
Total power savings (Kashgar) [10^6 kWh/y]	2,52	2,54
Power saving using customized rotors		
Power saving on summer operation [kW]	207	308
Power saving on average operation [kW]	320	279
Total power savings (Ashgabat) [10^6 kWh/y]	2,10	2,25
Total power savings (Kashgar) [10^6 kWh/y]	2,28	2,21

Table 3: Total power saving at different annual average loads, location and rotor design.

Investment costs for the cooling system as well as operational costs for water and electricity consumption are shown in Table 4. The price of water and electricity in both reference places are extremely low compared to the prices in Europe and USA. As a result, the operating costs are maintained at a very low level.

Ashgabat	Water	Power
Consumption [m3/month] and [kW/month]	2400	2880
Rate [Eur/m3] and [Eur/kW]	0,001	0,05
Operational cost [Eur/ month]	2,4	144
Capital cost [TEur]	160	
Kashgar	Water	Power
Consumption [m3/month] and [kW/month]	2300	2880
Rate [Eur/m3] and [Eur/kW]	0,02	0,06
Operational cost [Eur/month]	46	172,8
Capital cost [TEur]	160	

Table 4: Investment costs for passive cooling system and operational costs for water and power

Standard rotor design		
Average Main Air Compressor load	90%	100%
Energy savings (Ashgabat) [TEUR/year]	116.8	136.1
Energy savings (Kashgar) [TEUR/year]	151.2	152.2
Ammortization duration (Ashgabat) [year]	1.5	
Ammortization duration (Kashgar) [year]	1.1	
Customized rotor design		
Average Main Air Compressor load	90%	100%
Energy savings (Ashgabat) [TEUR/year]	104.9	112.1
Energy savings (Kashgar) [TEUR/year]	136.8	132.4
Ammortization duration (Ashgabat) [year]	1.6	
Ammortization duration (Kashgar) [year]	1.2	

Table 5: Results of the cost/benefit analysis

The results of the cost/benefit analysis are shown in Table 5. The investment costs can be amortized within 1.1 and 1.6 years, which is very attractive while considering the average investment life of machinery equipment for ASU application is more than 20 years.

MECHANICAL CONSIDERATIONS

The application of the passive cooling, in particular inlet fogging technique has also an impact on different mechanical issues as fouling, erosion and corrosion of the first stage of the compressor. Several studies in open literature [3, 8 and 9] have been focused on these issues. In general, the design of the fogging system has to take into account the impact of the increased humidity and eventual presence of droplets at the 1st impeller inlet. According to system manufacturers, the droplet size is in a range of about 15-20 µm and if the nozzles are placed sufficiently far from the impeller inlet, it is commonly assumed that all droplets will evaporate completely. However, it cannot be excluded that a certain (small) percentage of droplets will hit the impeller blades. Some CFD studies based on droplet models have shown that with this size the droplets will mainly follow the flow stream [10, 13 and 15].

A study is currently undergoing in order to assess the impact of this small percentage of droplets on the mechanical behaviour of the 1st impeller in RIKT compressors. It is anyway expected that such impact is not substantial. In fact, operating the RIKT with a very high humidity level is nothing exceptional since a large number of these compressors are functional without particular problems in industrial sites with tropical climate with almost constant 100% humidity during the rainy season.

In past years, some concerns have been raised about increased fouling when using a fogging system. It has been shown in few publications [8, 9] that the amount of fouling is not directly related to the increased humidity and it depends mainly on site conditions and the design of the inlet filter.

CONCLUSIONS

A comprehensive study concerning the effect of inlet cooling on isothermal inline (RIKT) centrifugal compressor has been carried out. This machine has the task to compress air from ambient conditions to until 6-7 bara to be further processed in the air separation process (ASU). Therefore, the compressor performance and the design are depending on the ambient conditions and seasonal weather site variations. By applying an inlet cooling, the maximum inlet volume flow achieved during hot days can be reduced. The advantage of such solution is that the compressor will not be oversized although taken into account high volume flow in hot days and therefore, it operates in average conditions in an optimized point.

Inlet cooling can be achieved by so called "passive" method (by injecting cooling water at ambient temperature with fogging or evaporative techniques) or actively cooling the coolant media. Unless there is a heat waste to recover, it results that the passive cooling is the most attractive technology since it requires very marginal additional cooling power (about 5 kW for cooling water pump and auxiliary). The advantages for the layout of the RIKT compressor come both from optimized stage design and optimal IGV settings in average conditions. In case of large seasonal variation and low humidity, the benefit in overall compression power is evaluated up to 1.0% at design conditions. In these cases, the thermo-economic analysis shows that the capital costs of a fogging system can be recovered in about 1.1 years of operation of the compressor, which is quite attractive given the average investment life of main air compressors for ASU applications being more than 20 years.

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NOMENCLATURE

ASU Air Separation Unit
COP Coefficient of performance
IGV Inlet Guide Vane
MAC Main Air Compressor
η_e Effectiveness of passive cooler
c_{pa} Specific heat capacity of the dry air
c_{pv} Specific heat capacity of the vapour
c_{pw} Specific heat capacity of the water
$h_{l+x,\infty}$ Enthalpy of atmospheric air
h_w Specific enthalpy of water
$h_{l+x,wb}$ Specific enthalpy of saturated air after passive cooling
$h_{l+x,c}$ Specific enthalpy of the air at the desired cooling temperature by active cooling
$\dot{m}_a$ Mass flow of the air before passive cooling
$\dot{m}_w$ Mass flow of the water
p_s Saturated vapour pressure
P_c Driving power by active cooling
$\dot{Q}_{tc}$ Cooling load by active cooling
r_0 Latent heat
T_∞ Ambient temperature
T_1 Compressor inlet air temperature after passive cooling
T_c Desired cooling temperature by active cooling
T_{wb} Wet bulb temperature
X_∞ Specific humidity of atmospheric air
X_1 Specific humidity of air after passive cooling
X_s Specific humidity of saturated air
X_c Specific humidity at the desired cooling temperature by active cooling

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